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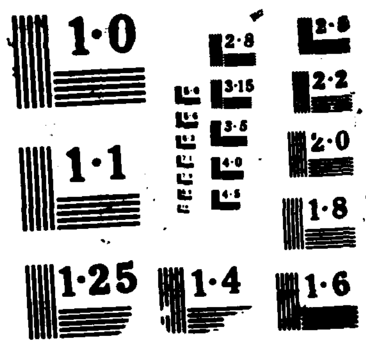
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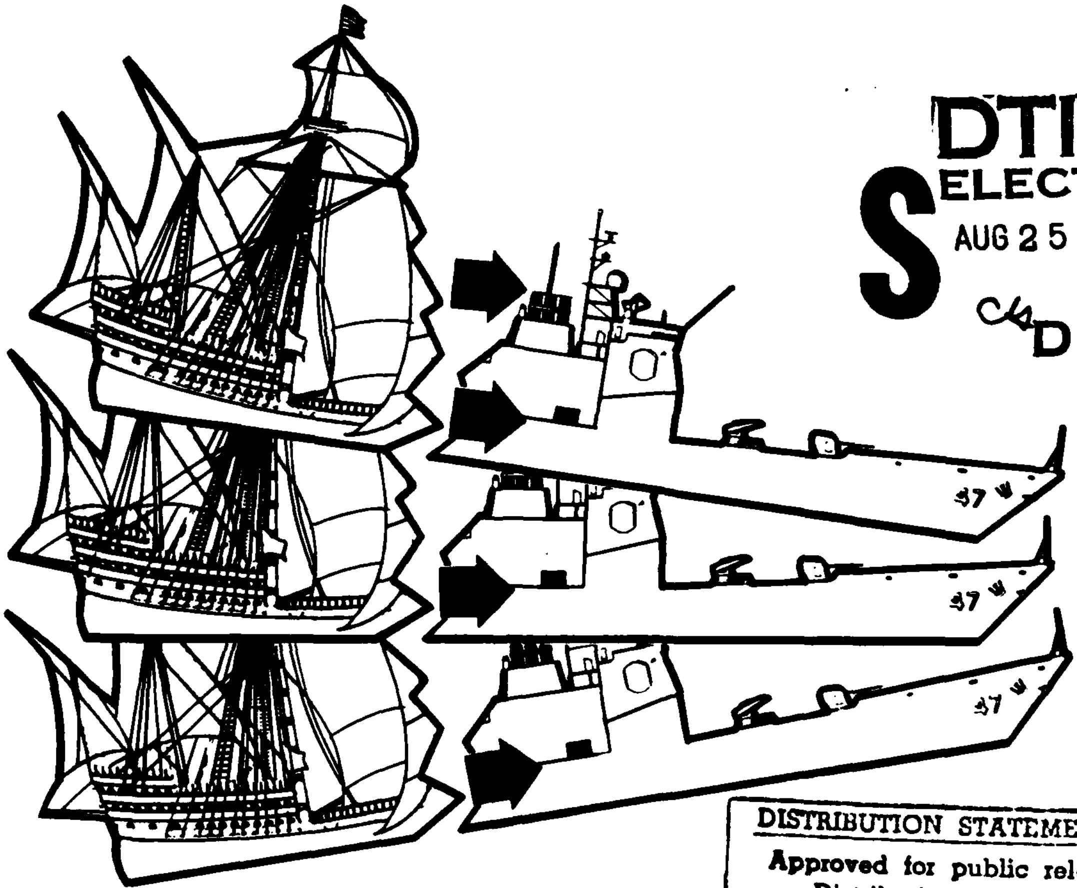

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DECISION RULES TO UTILIZE AN EXPERT SYSTEM FOR MACHINERY CONDITION ANALYSIS

by: Thomas A. Rutter

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DECISION RULES
TO
UTILIZE AN 'EXPERT SYSTEM'
FOR
MACHINERY CONDITION ANALYSIS

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ABSTRACT

Automated screening of machinery problems and their effect on machinery health and life expectancy, is an achievable and cost effective goal. "Expert" computer systems with automatic screening, and operator interface will enhance this decision making process.

This discussion is presented in three parts;

(1) the Aircraft Carrier experience in Machinery Condition Analysis, ~~our~~ development of a "smart" computer aided analysis system and our proposed "expert" system,

(2) a set of hierarchical production rules for machinery vibration analysis; and

(3) a tabulated Vibration Diagnostic Guide.

KEYWORDS

Patterns; structureborne; noise, vibration, acceleration, velocity, machinery condition, machinery health, shaft rate series; rotating machinery, residual imbalance; mass imbalance; couple, machine resonance; system resonance; matrix; history; trending, spectra, average files, baseline, continuum, cepstrum; combing.

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Subject

EXPERT SYSTEMS AND MACHINERY CONDITION ANALYSIS

BACKGROUND NOTES

To help develop the machinery repair package, we perform a machinery condition survey on approximately 390 machines for each of fifteen (15) aircraft carriers every nine to fourteen months. At PERA CV, we review recorded data on approximately 4800 machines (520 different types) each year. Our automated screening program compares the present data to the previous survey data and the average data plus one sigma for each machine type. The screening program will analyze at least ten major rotating parameters and the baseline continua for all frequencies thru 10,000 Hertz for each machine. All data is in 400 line spectra, order normalized and averaged - approximately 69,000 individual graphs per year.

Our major decisions involve: identifying a problem, determining mechanical severity, and judging the impact on machine performance and expected life. This is translated into determining the need for repair, the immediacy of repair required and whether to recommend a specific repair, or general overhaul of the entire unit.

The application of machinery condition analysis principles, particularly machinery noise and vibration analysis, appears to be a relatively simple, easily applied engineering discipline. There are many difficulties encountered in data acquisition, storage, and reduction, often in a harsh environment. However, the data acquisition phase approaches a science since the data can be repeated with some small calculable error. The analysis of that data is an art based on education, experience, and an intuitive feel for machine operation in relation to the recorded data. Each newcomer seems to have to develop his own feeling for machinery noise and vibration analysis - to "reinvent his own wheel". User friendly computerized analysis equipment only compounds the problem. (1)

Two problems that are becoming increasingly significant are aliasing problems and negative frequencies. With the newer digital analyzers, with variable aliasing filters, interference can be very subtle. (2) Due to the impulsive nature of rotating machinery, the resultant phasing errors, and the extremely narrowband capability of the modern real time analyzer, considerable care must be exercised. Quite often, the narrowband presentation will "average out" signals which are not stationary enough to remain in the filter bandwidth. Bearing tones are notable examples, as the development of bearing tones depends on bearing geometry, including uniform rotation of the bearing components. Consider the case of dried bearing grease restricting the motion of the cage. (3) Bearing noise will be audible. However, the cage will not rotate uniformly and the bearing tones will wander in frequency. The limitations imposed by the analyzer bandwidth or averaging technique may not faithfully reproduce such tones.

WE PRESENTLY USE A "SMART SYSTEM"
Our present system is relatively smart. Each evaluated rotating parameter is given it's own ranking factor, i.e., degree of importance based on previous engineering/operating experience. The system is "smart" since it has been supplied with the criteria and formulae to evaluate each parameter. The analyst can re-program but does not directly interface with the computer. The output is a ranking for each machine type (automatic screening) and a summation of all machines on that survey (condition index).

AN IMPROVED SYSTEM
Our proposed next generation of computer aided analysis will be an "expert system" allowing the computer to number crunch and the analyst to interface. The computer would have a series of decision trees. Such trees are presently available as commercial software packages, with the user supplying the appropriate line of reasoning and applicable data.

For an automated expert system to be useful, it should be able to assimilate what the analysts know. The system should understand the points of departure among the analysts who disagree, update it's corporate knowledge through questions and experience, and present it's reasoning in a usable manner.

This next generation will consist of a "knowledge base", which is mostly empirical, and an "imperfect knowledge" representation, consisting of rules which are assigned a probability factor. Experts in the particular problem domain are asked to assign these values (hence the term "expert system"). The "production rules" are basically an "if - then" structure. This then is a "matching path - determined search" system, which involves comparison of findings to those indicated in the "problem frame" and then selecting a set of frames that covers all the findings.

Fuzzy logic or an imperfect knowledge evaluation should result in a particular solution with considerable confidence.

The analysis should branch through a logic tree in several distinct steps:

1) Identify the shaft rate and determine if a clear harmonic series exists,
2) Determine if extraneous signals exist (i.e., bearing tones, cavitation, or an un-cataloged major rotating parameter), and evaluate their significance (see step 6,7 below)
3) Then, ignoring signals that are not exact multiples of the shaft rate, determine the pattern represented by the peaks of the shaft rate series,
4) Compare the magnitude of the signals to any criteria which may have been imposed,
5) Compare the individual shaft rate signals and patterns to measurements for other bearing caps on that machine,

and if available, to data from similar machines.

6) Comb the harmonic signals out of the data and smooth the baseline continuum,
7) and finally compare the baseline continuum to the previous and the averaged data to determine if the pickup locations are the same and if the system background noise has changed. This step will also help define the extraneous signals in step 2 above.

PROPOSED NEW DECISION RULES

Machinery noise and vibration should be considered as basically impulsive in nature. The shaft rate signal is quasi-sinusoidal and displays considerable stationarity. With severe imbalance, the shaft motion approaches a square wave. As the balance is improved, the shaft motion becomes more sinusoidal, and as the balance is further refined, the effect of the nonlinearity due to pedestal stiffness and lubrication dampening will become more evident. This distortion of the shaft rate signal generates a harmonic series (nF_r). A curve faired through the peaks of the shaft rate harmonic series will emphasize the distinctive patterns. A study of the application of shaft rate series pattern recognition to machinery condition analysis was presented to the National Bureau of Standards, Machinery Failure Prevention Group Symposium, Sept 1982. (5)

Triaxial measurements are required on each accessible bearing cap. The measurement directions are spoken of as: radial, transverse, and axial. The transverse direction is not a radial measurement, but a tangential measurement made on a small block on the periphery of the machine as close to a bearing cap as possible. The data should be recorded in velocity or acceleration and the frequency displayed should encompass at least 10 harmonics of the shaft rate - in logarithimic amplitude and linear freguency.

With one measurement point, we can determine frequency domain and time domain analysis. With two data points, we can do frequency domain, time domain and phase relationships. With the right two data points and a tachometer signal, we can do orbit studies. For indepth studies, one really should have triaxial measurements at all available bearing caps, plus a trackable shaft related signal or a tachometer signal.

With these thoughts in mind we have developed some decision rules and a tabulated Vibration Diagonistic Guide. They are presented as Appendix 'A' and 'B'.

SOME ADDITIONAL THOUGHTS

Vibration severity, overhaul criteria, severity charts, and maintenance goals offer an interesting insight into the thought process of the analist. Numerous authorities have attempted to develop a "Vibration Severity Chart", e.g., Rathborn, Blake, IRD, Mil Std 167, Mil Std 740b, etc. None have been particularly

successful even when limited to a specific parameter, e.g., imbalance. The application requires considerable engineering insight. Imposition of a pass/fail criteria does not consider the inherent variability of structureborne noise and vibration data. In each case the most important thought is to be aware of the type of machinery the individual was working with when that chart was developed, and his reason for developing that chart. Much of the early work was for insurance liability.

The most successful "maintenance goals" have been comparison to similar machines, specifically to averaged data for several similar machines. "We have achieved these levels before and should be able to return a machine to the same condition."

As a maintenance goal, we much prefer to use the averages of several similar machines plus a standard deviation. One good application of computer manipulation of large volumes of vibration data is in the development of averaged data and deviation from the average to provide a "maintenance goal".

Mil Std 167, which is the present standard for procurement of rotating equipment for our Naval Forces, should be considered a "failure criteria". That is, if a machine exceeds "167" at the shaft rate (balance), it has a limited life expectancy. Application of "167" to higher orders or frequencies is also very suspect. This is in line with the concept of well balanced and aligned machinery having shaft rate signals below one mil p/p. There is an effort underway in the Navy community to revise Mil Std 167. In the interim, the use of averaged and averaged plus one sigma data for a "maintenance goal" is highly recommended.

CONCLUSIONS

We have presented a series of hierarchical production rules and a knowledge base which is mostly empirical. These production rules, taken in the sequence shown, will provide an assessment of machinery health. These rules are the basic building blocks for the development of an "artificial intelligience" analysis system.

ACKNOWLEDGEMENTS

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APPENDIX 'A'

ANALYSIS NOTES AND HINTS

For this discussion we will assume the analyst is at least aware of the several advanced analysis techniques; especially the various correlation processes and Cepstrum (2), pattern recognition (5), and phase relationships (6). The analysis (software) programing must be capable of accepting multiple data sources and evaluating phase relationships, combing techniques, curve fitting, and smoothing. A Cepstrum analysis will help to define the harmonic series and would be included in the average information for automatic screening.

Machinery noise and vibration should be considered as basically impulsive in nature. The shaft rate signal is quasi-sinusoidal and displays considerable stationarity. With severe imbalance, the shaft motion approaches a square wave. As the balance is improved, the shaft motion becomes more sinusoidal, and as the balance is further refined, the effect of the nonlinearity due to pedestal stiffness and lubrication dampening will become more evident.

For a horizontal machine or for measurements made on the feet, the one-time shaft rate signal in the vertical direction will usually be the highest. The highest levels tend to be near the end or the component requiring balance.

A one-time shaft rate signal dominating in the axial direction will most likely be an indication of improper preload. There will be excessive clearance or looseness in the axial direction. Quite often this will be due to a component missing, such as a wavey washer. Other causes can be thrust bearing wear, irregular wear patterns on a ball bearing race, and also misaligned or cocked bearings which pinch the cage and forces the balls into an irregular pattern.

If there is a coupling misalignment involved, there should also be a strong two-time shaft rate signal in the radial direction. As the condition worsens, the two-times shaft rate signal in the axial direction will become stronger. Often one-half shaft rate signals may appear.

If the one-time shaft rate signal dominates in the transverse direction (over the radial and axial direction), then look for a broadening of the base of the signal and/or tones near/or by the shaft rate signal suggesting a resonant condition. For a vertical machine, a dominant transverse signal at the top (bearing cap) is usually due to a lack of stiffness in the transition assembly or foundation vice a resonance if there is no broadening of the base of the shaft rate signal. For a horizontal machine, if there is no broadening of the base of the shaft rate signal, then a dominating transverse shaft rate signal would suggest a lack of

stiffness in the pedestal. Remember, a lack of stiffness and resonance can occur at the same time. A rap test will help determine the actual resonant frequencies, but will not identify a lack of stiffness.

For a two-time shaft rate signal, dominating in the radial direction without a strong one-time shaft signal in the axial direction (often for a horizontal machine, the transverse will be higher than the vertical), this is a fairly good indication that it is not a coupling misalignment. It indicates some unit (the impeller, or the bearing) is cocked on the shaft. If it is the bearing, the signature will usually show bearing tones; if it is an impeller the data will usually show a vane rate series; and if it is the armature, there will be electrical signals present.

Returning to a one-time shaft rate signal dominating in the axial direction there is one peculiar problem and that is where the unit has a cantilevered component. That is, there is no bearing support on the free end. Examples which come to mind include: close coupled pumps and exhaust fans where the impeller is on the end of the shaft. An unusual case are ships service turbine generator overspeed trip mechanisms, where the overspeed trip mechanism is cantilevered off the end of the shaft. The basic problem is radial imbalance of the cantilevered component introducing a gyroscopic moment and coupling the deflection to the axial direction. After setting the overspeed trip by moving the trip weight, who rebalances the turbine?

A two-time shaft rate signal, dominating in the radial direction, (measured on both ends of the shaft, usually within 6 - 8 db of each other and 10 - 15 db higher than the related shaft rate signals), could be caused by a bent or bowed shaft (check phase to be sure). I've also noticed when sleeve bearings were loose, that is, when the shell was loose in the bearing housing, microbalancing the unit causes the shaft rate signal to go down, but the two-time shaft rate signal remained high.

For a strong shaft rate series that doesn't appear to dominate in any direction, the cause will usually be mechanical looseness in the bearings, e.g., clearance between the balls and the race. The vertical will usually be the highest for a horizontal machine and the series may extend through the 20th harmonic or higher. If one bearing cap exhibits a strong two-times and four-times shaft rate signal, look for a mechanical misalignment of that bearing or improper installation, that is the housing is cocked on the shaft or shoulder, e.g., dirt on the shaft or shoulder.

For a strong shaft rate series, where the levels suggest a shaft may be hammering; and for certain basically imbalanced machinery, (such as piston compressors), even harmonics, which dominate the shaft rate series pattern suggest misalignment, while odd harmonics suggest imbalance.

For IMO pumps, one must also consider the number of threads, that is, whether there are two grooves or a single groove on the idler assemblies. This will help determine the difference between misalignment and pump pulsations. Pump pulsations will be the number of idlers times the number of threads.

UNIQUE PROBLEMS FOR CENTRIFUGAL PUMPS

The concept of misalignment for close coupled or solid shafted machines may be difficult to visualize, but the foregoing analysis suggestions should offer some insight.

Two-time shaft rate signals can also occur due to the pump internals being uneven. If it does occur, there should also be a strong vane rate series. An oversized impeller that is not rubbing will also show up as high motor rotor amperage, and higher than normal pressure increase across the pump.

If the pump has not been vented and is being operated in standby or recirc mode, the turbine RPM will be low (typically 100 to 250 RPM), while for an electric motor driver the RPM will be higher than normal (typically less than 1% vice 2-3% slip). Driver imbalance (shaft rate) will be low while pump imbalance will be up. Vane rate signals will be up. Cavitation (air in the system) will be evident, with the vane passing frequency the dominant stationary component. A cavitation signature could be thought of as pseudo random noise with considerable stationarity.

PHASE MEASUREMENTS

At the shaft rate, $60-90^{0}$ between a radial and transverse measurement seems to be as good an indication of balance as 180^{0} between two axial measurements is of misalignment. (6)

To be sure of coupling shaft to shaft phase measurement, you must measure across the coupling itself. Measurements at opposite ends of a machine introduce misalignment signals due to other components, e.g., cocked bearings.

APPENDIX 'B'

THE VIBRATION DIAGONISTIC GUIDE

A tabulation of common and not so common mechanical problems and the prominent signs of their presence is presented. It is an attempt to put many years cumulative experience into a format a starting analyst could use. It will never be complete but should serve as a starting matrix for an artificial intelligence system and as a memory review for the analyst.

VIBRATION SOURCE	DOMINATING ORDER	DOMINANT PLANE	PHASE ANGLE RELATION	AMPLITUDE	ENVELOPE CHARACTERISTIC	COMMENTS
VIBRATION TYPE: IMBALANCE (NOTE 1)						
1.) Mass Imbalance, Simply Supported						
a.) Forced (Static)	1X (NOTE 2)	radial (NOTE 3)	in phase	steady	all narrow band (NOTE 4, 5)	1) in phase radial to adjacen radial and except for coup 90° radial to transverse
b.) Couple	1X	radial	180° out of phase			
c.) Quasi-Static	1X	radial	in phase			2) Quasi-static and dynamic most common
d.) Dynamic	1X	radial	out of phase (random)			
2.) Mass Imbalance (Cantilevered)	1X	axial	60-90° radial to transverse			
3.) Bent and Bowed Shaft	1X(usually does not dominate)	axial	if no coupling, (1X) 180° out of phase axially from end bell to end bell			1) Rotor bow due to thermal stresses may cause change in (1X) amplitude and phase with time
	2X	radial	(see notes and hints)			2) Runout at the rotor mass should appear as imbalanc
4.) Resonance/Lack of Stiffness	1X	transverse	(60-90° radial to transverse)	varying	narrow band with broad base	1) (see notes and hints)
VIBRATION TYPE: MISALIGNMENT						
1.) Improper Preload	1X	axial	in phase axial to axial and (60-90° radial to transverse)	steady		(see notes and hints)
2.) Shaft to Shaft						
a.) Parallelism	2X-(1X)	radial	1X 180° out of phase between two adjacent radials	steady	all narrow band	1) Runout at the rotor mass should appear as imbalance (1X) 2) Most misalignment will be a combination of parallelism and angularity
b.) Angularity	1X-(2X)	axial	1X 180° out of phase between two adjacent axials			3) On long coupling spans 1X will be higher 4) for strong odd or even harmonics or for IMO pumps 5) misalignment errors in the vertical plane are most common
3.) Components Misaligned to Shaft Centerline	2X	radial	1X 60-90° radial to transverse	steady	narrowband	1) (see notes and hints)

VIBRATION SOURCE	DOMINATING ORDER	DOMINANT PLANE	PHASE ANGLE RELATION	AMPLITUDE	ENVELOPE CHARACTERISTIC	COMMENTS
VIBRATION TYPE: ELECTRICALLY INDUCED						
1.) Eccentric Motor Rotor	1X (imbalance)	radial	60-90^0 radial to transverse		narrow band	
	line freq. and multiples	any direction		fluctuating (see notes and hints)	narrow band with side bands usually 2x line freq.	Line freq. multiples will fluctuate (beat) in amplitude
2.) Loose Stator Laminations	2x line freq.	any direction		steady	narrow band	
	and high freq. (over 1000 Hz) (Note 2)			high, steady	narrow band (with sidebands usually 2X line frequency)	Not usually destructive
3.) Broken Rotor Bar	rotor bar rate (35 to 80X)	any direction		steady	narrow band of energy	Only occurs on induction motors
	shaft harmonics with slip freq. sidebands	any direction		steady		Slip freq. = line freq. minus rotation speed.
4.) Unbalanced Line Voltage	1x,2x line freq.	any direction (usually radial strongest)		low, steady	narrow band	High current and excessive heat
5.) Local Stator Heating	2x line freq. (with 2x slip, sidebands)	any direction	N/A	increasing with temperature (see notes and hints)	narrow band with sidebands	1) Heating may be caused by shorted laminations 2) As motor heats up, vibrati may get worse.
6.) Local Rotor Heating	1x, 2x line freq. 2x slip freq. (Also 2x slip sidebands on 1X or 2x line freq.)	radial	N/A	increasing with temperature (see notes and hints)		1) Heating may be caused by lamination problems, broken or cracked rotor bar, or poor rotor/shaft contact 2) As rotor heats up, vibrati may get worse.
VIBRATION TYPE: MECHANICAL PUMP PROBLEMS						
1.) Non-rotating Looseness	2X	radial	varies	steady	narrowband	
2.) Rotating Looseness (Rotors, Impellors) (see also imbalance)	1X dominant	radial	varies (usually 60 to 90^0 radial to transverse)	will vary from start-up to start-up		1) Variance of amplitude or phase may be caused by center of gravity shifts.
3.) Rotor Rub (see also subharmonic shaft related signals)	0.5X,1X	radial	N/A	1) 0.5X steady 2) 20 to 35X 1X (or higher in severe cases)	narrowband broadband	1) May excite rotor resonance

VIBRATION SOURCE	DOMINATING ORDER	DOMINANT PLANE	PHASE ANGLE RELATION	AMPLITUDE	ENVELOPE CHARACTERISTIC	COMMENTS
VIBRATION TYPE: HYDRAULIC PUMP PROBLEMS						
1.) Centrifugal Pump with (n) Vanes	(n)X and harmonics	radial (predominantly in the direction of discharge piping)	N/A	fluctuating	narrowband (hydraulic resonance will broaden signal base)	More than one (1) discharge volute (as in multi-rotor pumps) will produce harmonics of blade rate
2.) Gear Pump with (n) teeth	(n)X,2(n)X	radial	N/A	steady	narrowband	
3.) Cavitation or Starvation	Random noise		N/A	fluctuating	broadband to 2000 + Hz	
4.) Screw Pumps (IMO)						(see notes and hints)
VIBRATION TYPE: DRIVE BELT PROBLEMS						
1.) Mismatched, worn or stretched belts	Muliples of belt freq. (2X belt freq. usually dominant)	radial, in line with belts	N/A	may be unsteady if belt frequency close to either shaft speed	narrowband - unsteady with side bands	For all belt "slap" problems confirm with strobe light or belt exitation techniques
2.) Eccentric and/or imbalanced sheaves	1X shaft	radial	(see also imbalance)	steady	narrowband	
3.) Drive-belt or sheave face misalignment	2X shaft	axial	in-phase	steady		
	2(n)X	radial	N/A	unsteady		
4.) Drive belt resonance	Belt resonance with no relationship to rotational speed	radial	N/A	may be unsteady	broadband	20% of resonant frequency depending on dampening
VIBRATION TYPE: GEAR PROBLEMS						
1.) Gear Mesh Errors with (n) Teeth	(n)X and harmonics	Radial for spur gears	N/A	depends on loading, speed and error	Usually single peak	
		Axial and radial for single or double helical gears			Sometimes sidebands	
2.) Pitch line runout, Mass imbalance Misalignment or Faulty Tooth	1X and gear mesh freq. with +gear rotatIng freq. sidebands	Radial for spur gears	N/A	steady	discrete peaks	May excite transverse or torsional resonances at various frequencies
		Axial for single or double helical gears				
3.) Hobbing Error	≅ 300X	N/A	N/A	steady	narrow band	Severe machining errors due to hobbing may also cause high 2x or 3x gear rotating frequency vibration

NOTE: Machining errors (precession) every third or every second tooth will cause 1/3 or 1/2 of expected gear mesh signals.

VIBRATION SOURCE	DOMINATING ORDER	DOMINANT PLANE	PHASE ANGLE RELATION	AMPLITUDE	ENVELOPE CHARACTERISTIC	COMMENTS
VIBRATION TYPE: SLEEVE TYPE BEARINGS						
1.) Large Bearing Clearance	A hump in the ascending shaft rate series at 4 to 8X and/or 7 to 15X (3)	radial	N/A			(see notes and hints)
2.) Bearing Rub Hysteritic Whirl Torque Rub	0.5X	radial	N/A			
3.) Oil Whirl	less than 0.48X	radial	N/A			
4.) Oil Whip	shaft critical $\cong$ 0.65X	radial				Oil whirl infers precession of the shaft. Oil whip infers planar motion usually associated with a shaft critical (resonance)
VIBRATION TYPE: DEFECTIVE BALL BEARINGS (Outer Race Stationary)						
1.) f_r = rotating frequency = RPM/60 = X f_t = train frequency = .38 to .42X (Note, unstable) f_s = ball spin freq. = 1.5 to 3.0X (Note, usually 2x ball spin will be evident first.) f_{ir} = irregularity freq. of inner race = 4 to 10X f_{is} = irregularity freq. of outer race = 2 to 7X (Ref 2)				varying	narrow band	
NOTE: Exact bearing tones can be calculated (Ref 4). The frequencies of bearing tones may become lower as the bearing wears and balls spin slower - slide more - (i.e., dry grease or pinched cage)						
2.) Inadequate Lubrication		radial	N/A		broad band	Narrow band natural frequency of bearing rings can be estimated from pipe resonances formulas
3.) Large Bearing Clearance	A hump in the ascending shaft rate series at 4 to 8X or 7 to 15X (Ref 5)	radial	N/A		narrow band	
4.) Heavy Thrust Load	ball spin.	axial	N/A		narrow band	
5.) Cocked Bearing (See also lack of preload and analysis notes)	2X	radial and/or tangential	N/A	high, steady	narrow band	Accompanied by low axial amplitudes

NOTES

1) The basic chart format was adopted from a proposed diagnostic chart by DLI Engineering Corporation (Bert Lundgaard), Bainbridge Island, WA.
2) 'X' means order of multiple of rotational frequency (f_r) where f_r = RPM/60, and 'x' means multiplication.
3) Radial, Transverse and Axial mean transducer orientation referred to the shaft
4) Broad band signals may also be narrow bands consisting of tones with side bands or modulation products.
5) Narrow band infers a single frequency. Broad band or wide band refers to any signal not a single frequency. Since this will be a function of analyzer band width, the lower analysis ranges are usually referred to as narrow bands of energy.

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